Thermal management with micro-architectured materials and metal foams

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Sponsors: ONR, ONR IFO, EPSRC, and Porvair Ltd.

Outline

- **Utilisation of micro-architectured structure as a heat sink medium.**

1. Analytical approach using two-equation model.

2. Experimental results of pressure drop and heat transfer measurements

- **Novel lightweight metal foams**

1. Examples employing metal foams as a heat exchanger

2. Analytical approach using two-equation model.

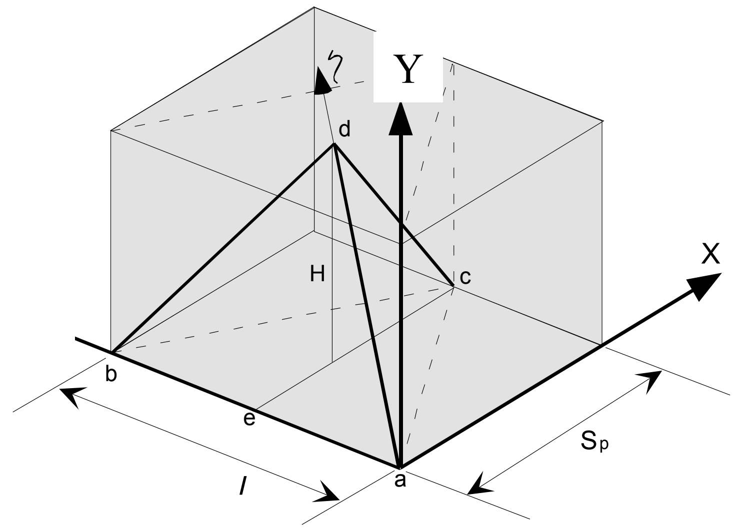
3. Experimental results of pressure drop and heat transfer measurements

Heat sink model using the LFMs

Lattice-frame material (LFM)s are triangulated 3-D structures (Tetrahedral cell based)

Greater lateral stiffness
and lower cost per unit weight over honeycombs and metal foams

- Surface area density : 123.68 [1/m]
- Relative density (or porosity) : 0.062 (0.938)

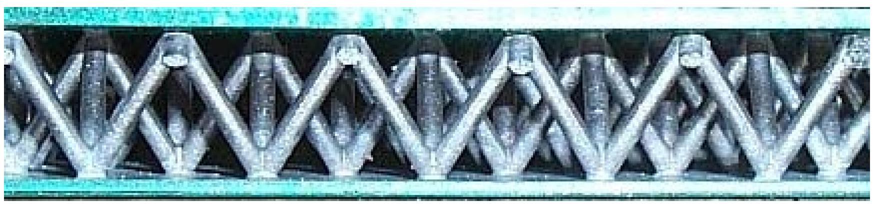


Unit cell of the LFMs

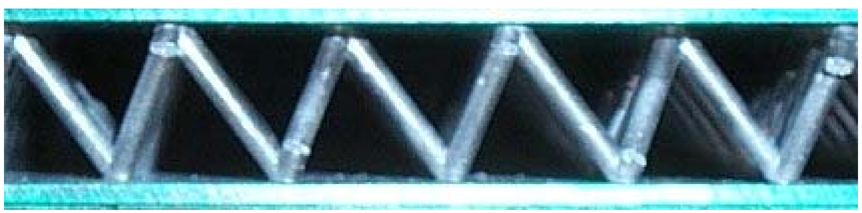
Forced air convection heat transfer measurements were conducted

Configuration of Lattice-Frame Materials

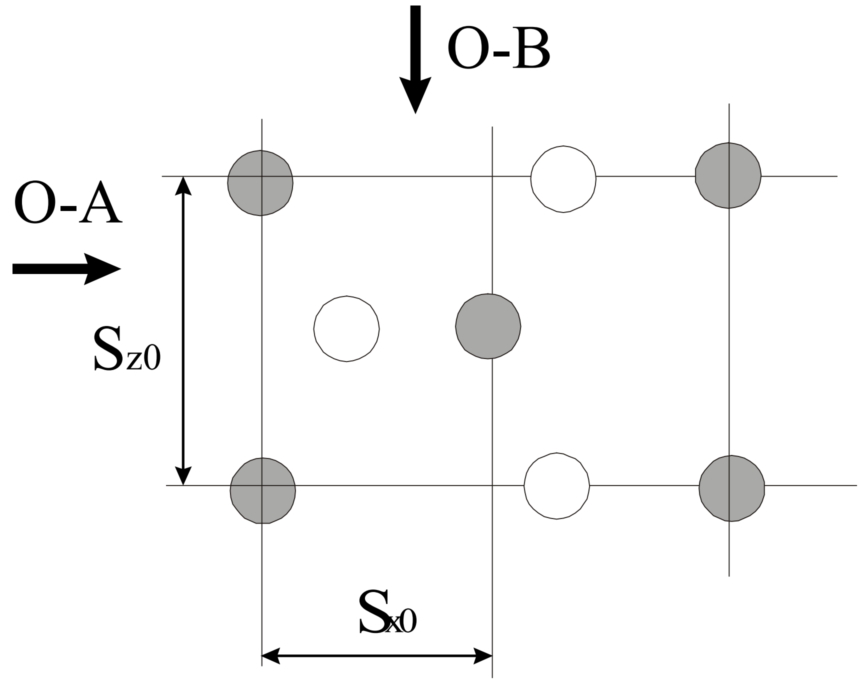
Two orientations denoted as below were tested for heat transfer and pressure drop measurements



(a) Orientation-A (frontal view)



(b) Orientation-B (frontal view)

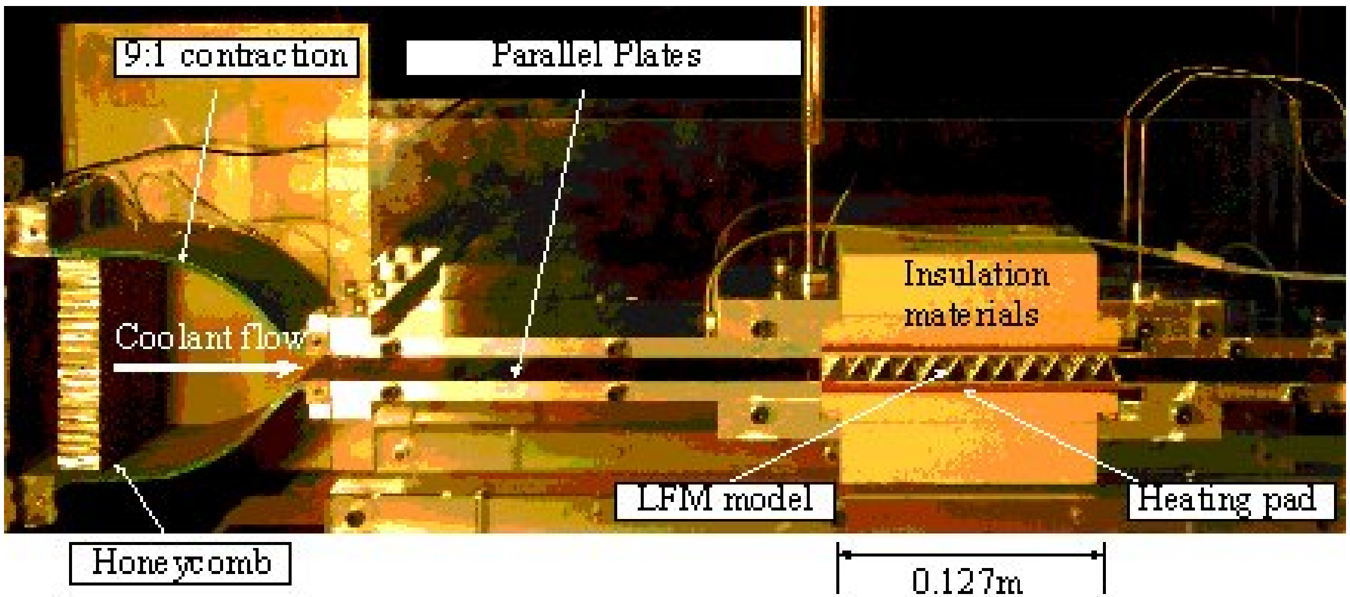


(c) Crossflow arrangement of LFMs

Experimental conditions & set-up

Specification of LFM		Typical operating parameter	
LFM cell bar diameter (*d*)	0.002 m	Inlet coolant mean velocity	1.0~26.0 m/s
Longitudinal cell pitch (S_{X0})	0.0127 m	Bar Reynolds number	$120 \leq \mathrm{Re}_d \leq 3200$
Transverse cell pitch (S_{Z0})	0.0147 m	Input heat flux	4, 8 and 16 KW/m^2
Cell stut length (*l*)	0.0147 m	Inlet coolant temperature	300 K
Cell height (*H*)	0.012 m	Outlet coolant temperature	305.0 K~360.0 K
Material	LM25	Test section inlet pressure	1 bar (ambient condition)

Table of parameters for the current test



Photograph of test rig and inserted LFM test sample

Governing Equations

Momentum Equation

$$0 = -\frac{d}{dx}\langle p\rangle_f + \frac{\mu_f}{\varepsilon}\frac{d^2}{dy^2}\langle u\rangle - \frac{\mu_f}{K}\langle u\rangle - \frac{\rho_f F}{\sqrt{K}}\langle u\rangle^2$$

Energy Equation

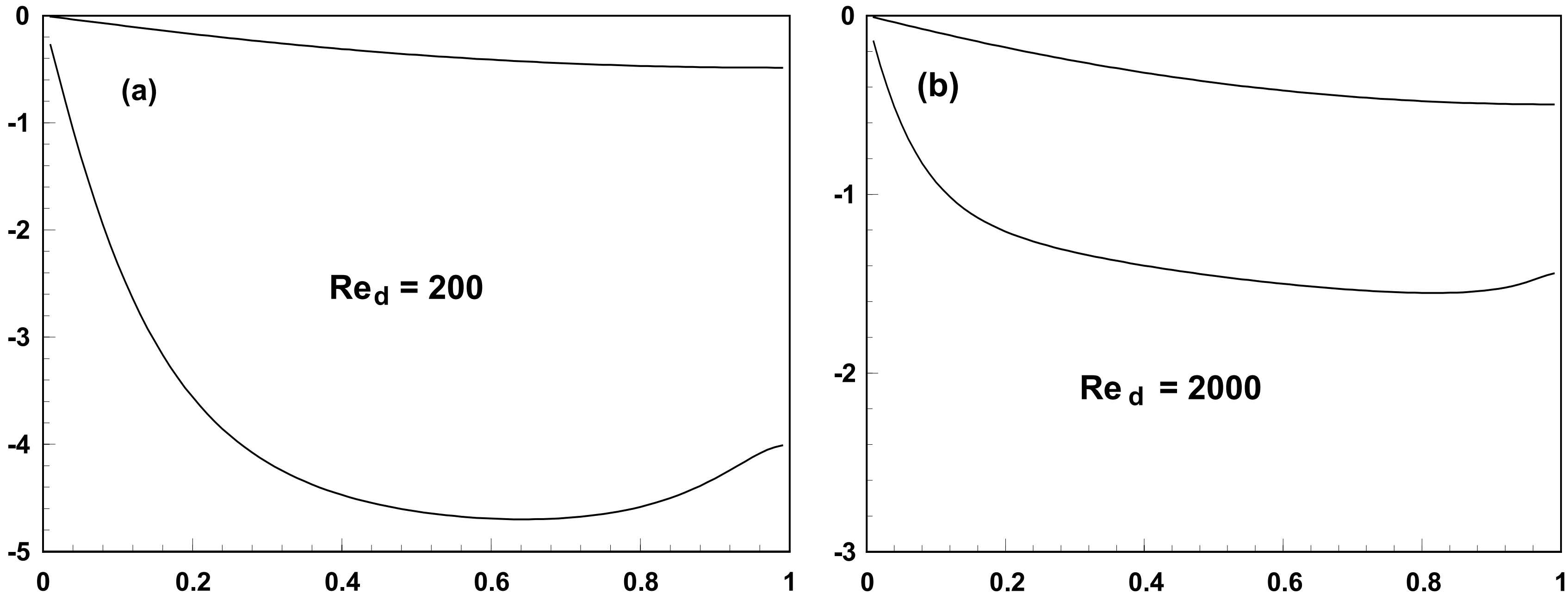
$$k_{se}\frac{\partial^2\langle T\rangle_s}{\partial y^2} = h\tilde{a}\left(\langle T\rangle_s - \langle T\rangle_f\right)$$

$$\rho_f C_f \langle u\rangle_f \frac{\partial\langle T\rangle_f}{\partial x} = h\tilde{a}\left(\langle T\rangle_s - \langle T\rangle_f\right) + k_{fe}\frac{\partial^2\langle T\rangle_f}{\partial y^2}$$

K = permeability
ρ_f = fluid density
F = inertial coefficient
ε = porosity
wetted area ratio
k_{se}, k_{fe} = effective thermal conductivity
C_f = specific heat of fluid
h = interstitial heat transfer coefficient;

Constant heat flux
Flow and heat transfer are fully developed

$$\theta_s = \frac{\langle T \rangle_s - \langle T \rangle_w}{q_w H / k_{se}} \quad \theta_f = \frac{\langle T \rangle_f - \langle T \rangle_w}{q_w H / k_{se}} \qquad \mathrm{Re}_d = \frac{u_m d}{\nu} \qquad Y = \frac{y}{H}$$



Solid and fluid temperature distributions for different Reynolds numbers

Results of Hydraulic Resistance

- Friction Factor

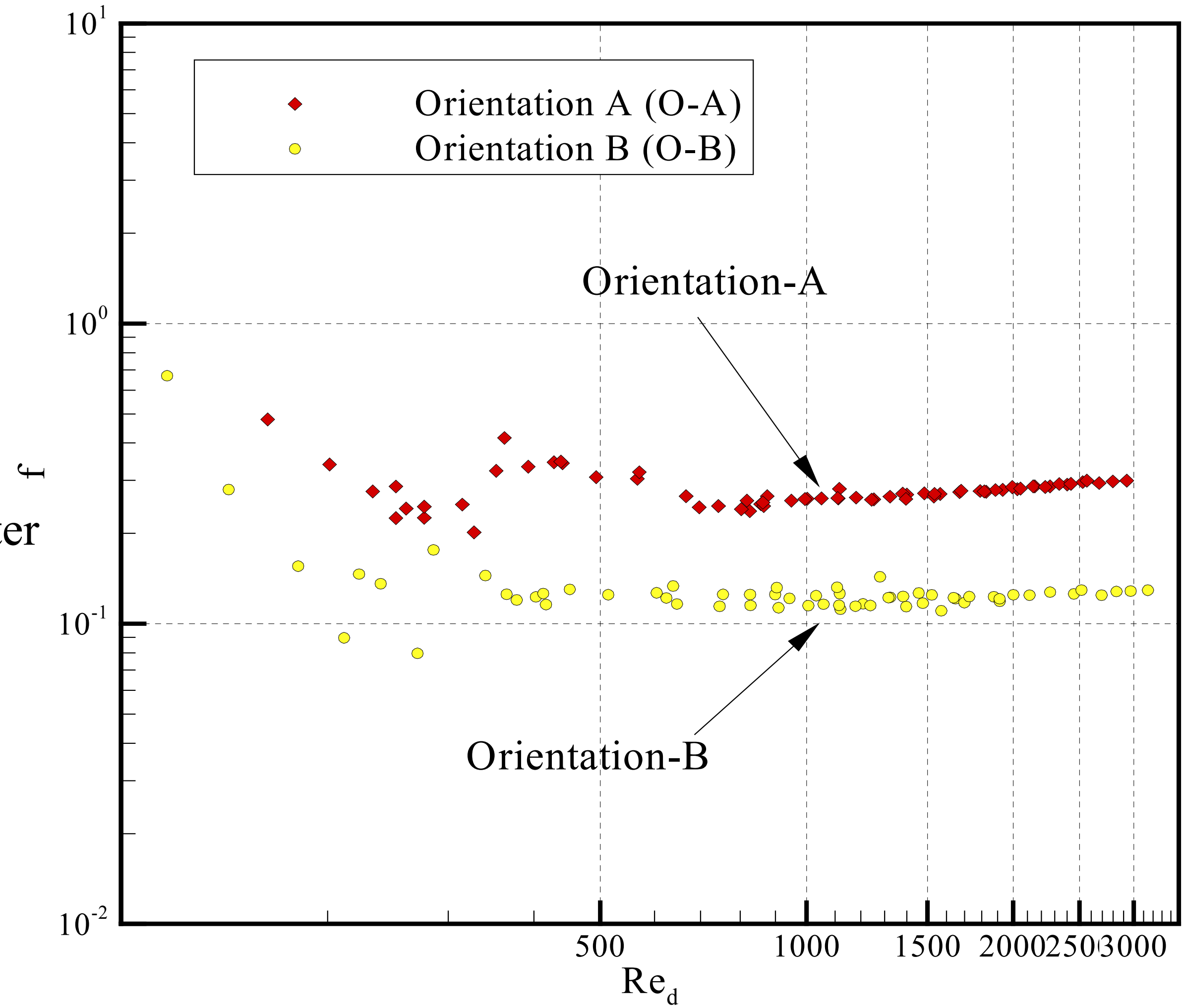
$$f = \frac{\Delta P}{L} \frac{D_h}{4(\rho U_m^2 / 2)}$$

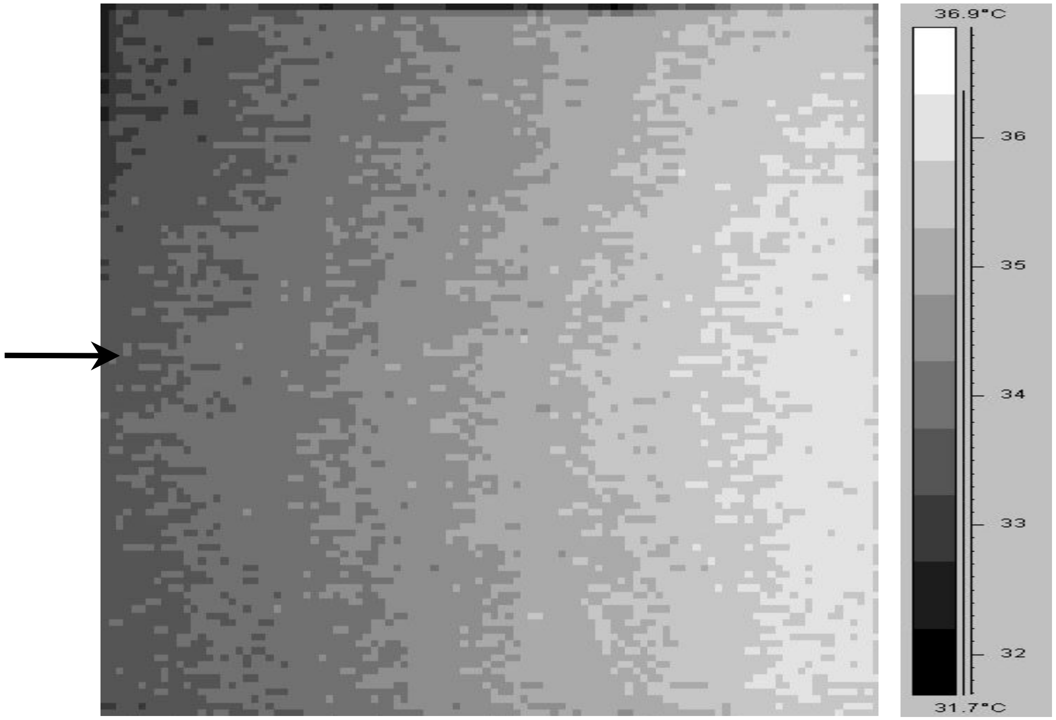
-Reynolds number
based on LFM cell bar diameter

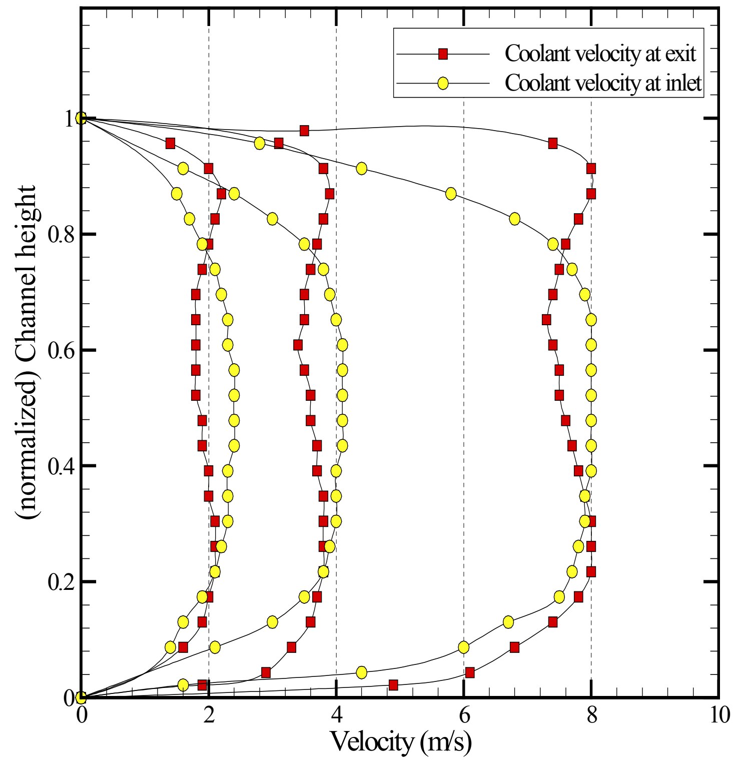
$$Re_d = \rho\ Ud\ / \mu$$

-Operating range

$$120 \le Re_d \le 3200$$



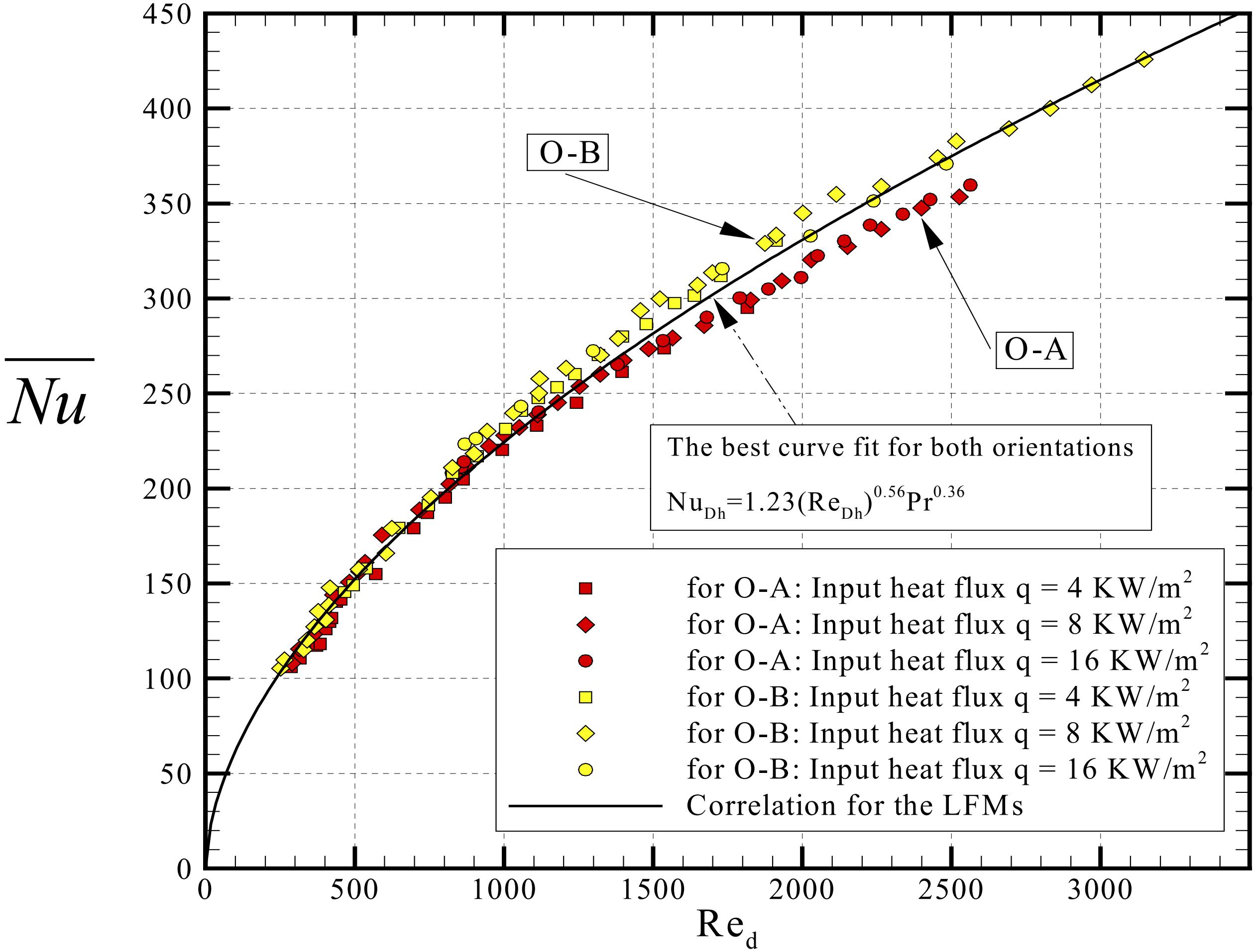




Infrared thermal image of top LFM substrate showing linear increase of temperature in constant heat flux boundary condition

Coolant velocity distributions along the channel height for both inlet and outlet of the test section

Average Nusselt number



Average temperature difference

$$\Delta T_{avg} = \left(\sum_{i=1}^{n} T_{w,i}\right)/n - T_{in}$$

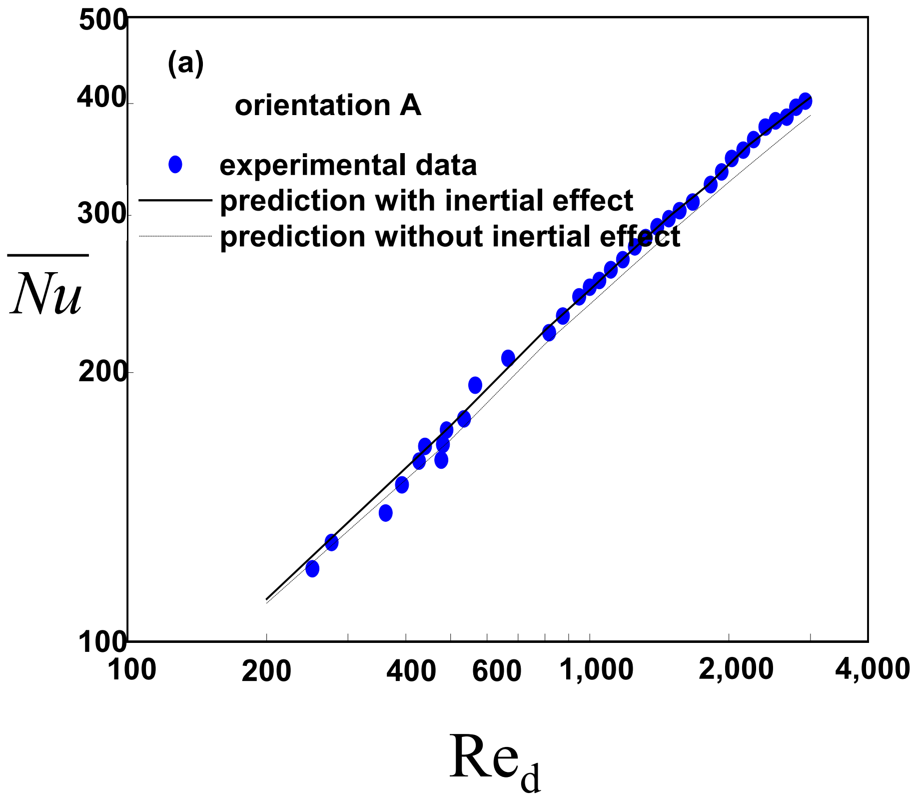
Heat transfer coefficient

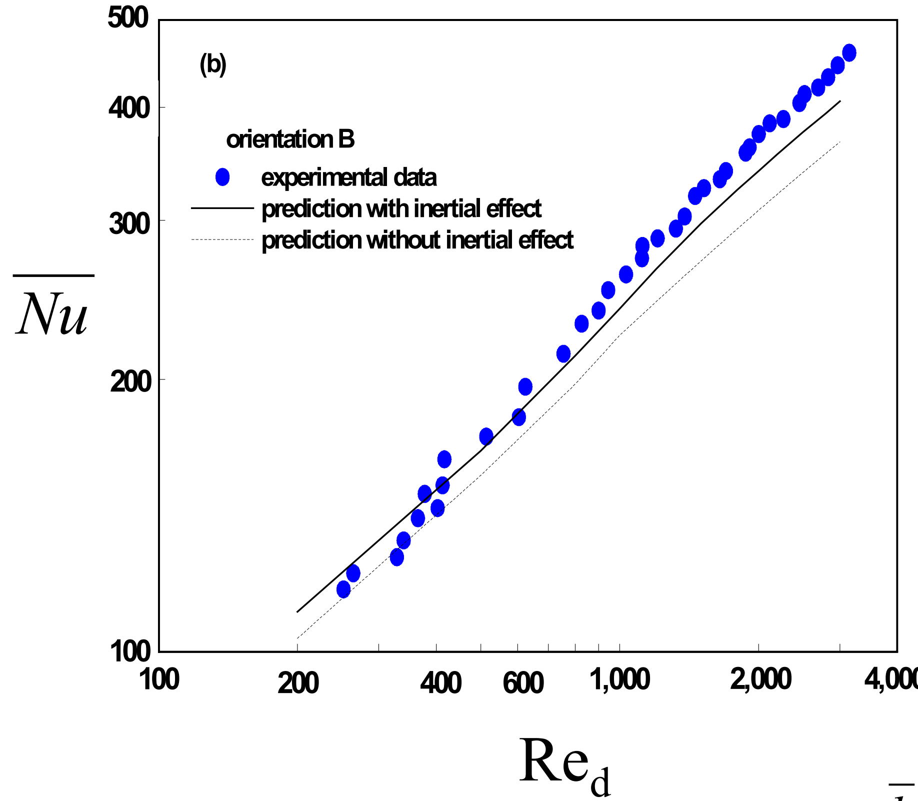
$$\bar{h} = q/\Delta T_{avg}$$

Average Nusselt number

$$\overline{Nu} = \bar{h}D_h/k_f$$

Comparisons



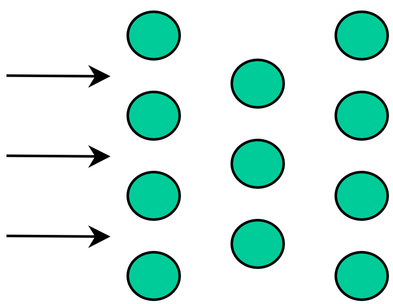


Comparison of overall Nusselt number with experimental data

$$\bar{h} = \frac{q}{T_w - T_{f,b}} \qquad \overline{Nu} = \frac{\bar{h} D_h}{k_f}$$

Interstitial heat transfer coefficient, h proposed by Zukauskas for Staggered banks of cylinder array

Staggered cylinders



$$h' = 1.04\,\mathrm{Re}_d^{0.4}\,\mathrm{Pr}^{0.36}\,\frac{k_f}{d} \qquad \mathrm{Re_d} = 1 - 500$$

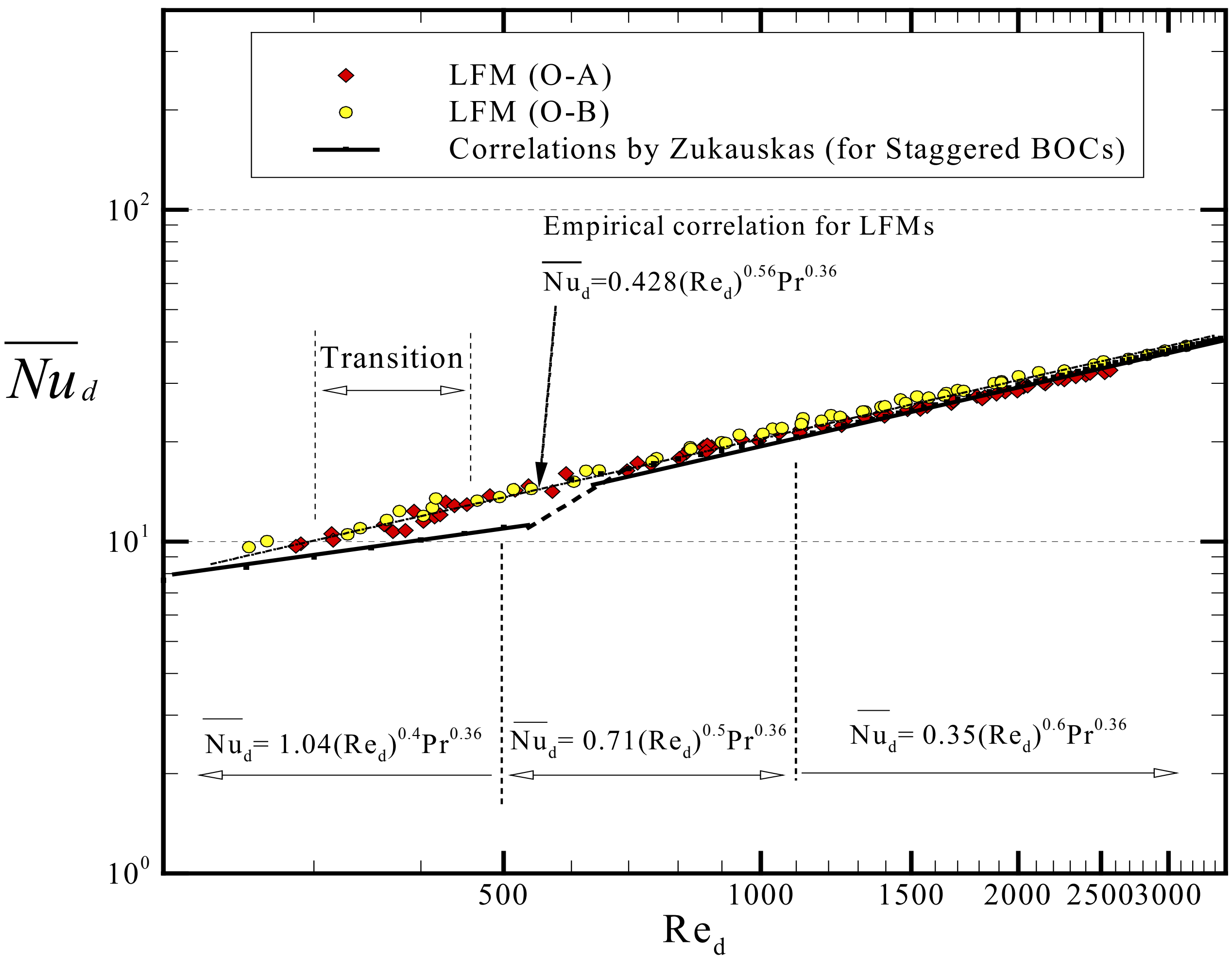
$$= 0.71\,\mathrm{Re}_d^{0.5}\,\mathrm{Pr}^{0.36}\,\frac{k_f}{d} \qquad = 500 - 1000$$

$$= 0.35\,\mathrm{Re}_d^{0.6}\,\mathrm{Pr}^{0.36}\,\frac{k_f}{d} \qquad = 10^3 - 2\times10^5$$

Effect of yaw angle

$$h = 0.9h'$$

Average Nusselt number



Reynolds number based on cylinder bar diameter used. where

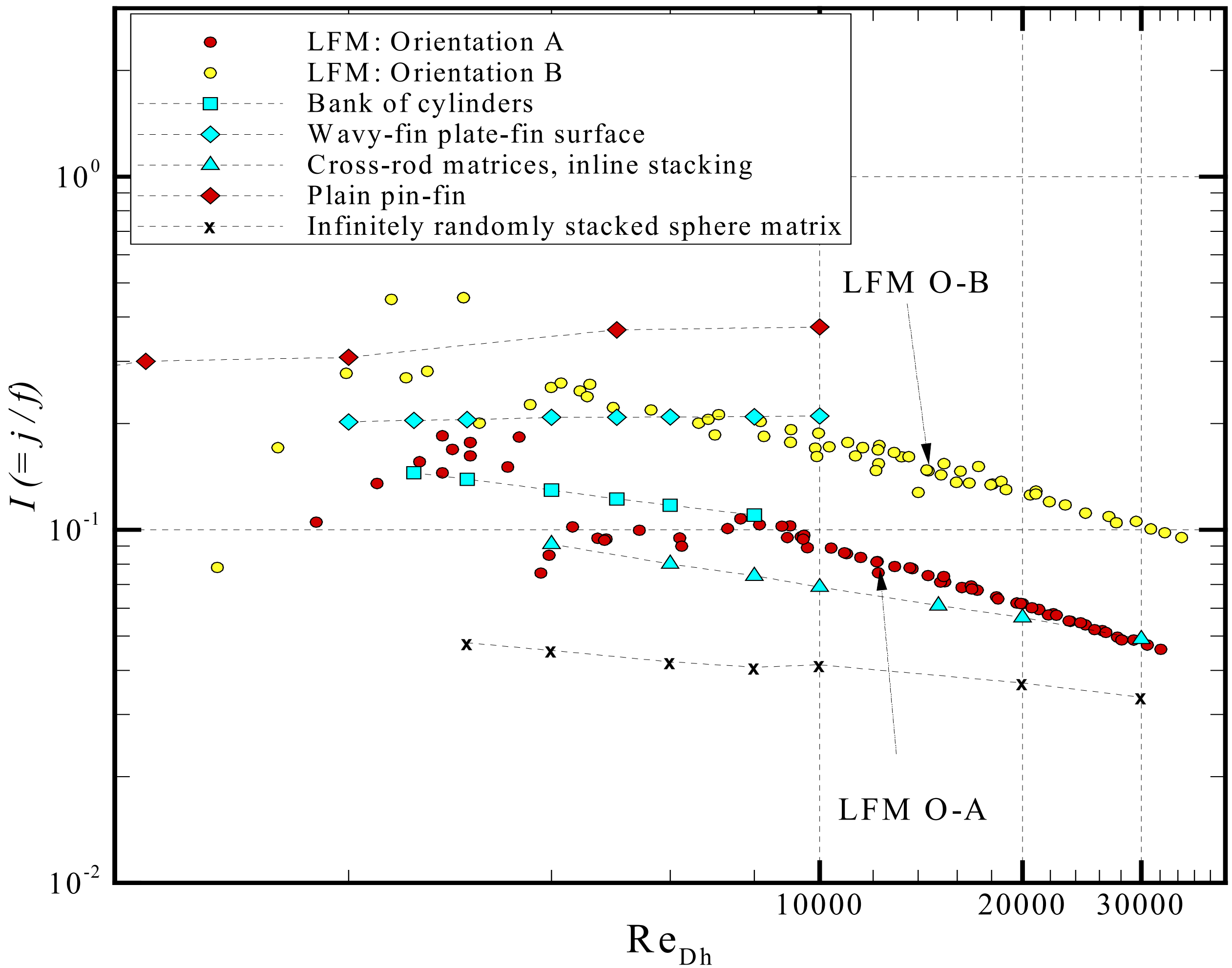
$$\overline{Nu}_d = \frac{\overline{h}}{k_f / d}$$

$$Re_d = U_m d / \nu$$

and Prandtl number Pr is assumed to be constant e.g. 0.71

Comparisons with other heat sink media

(from Kays and London, "Compact heat exchangers")



J-Colburn factor :

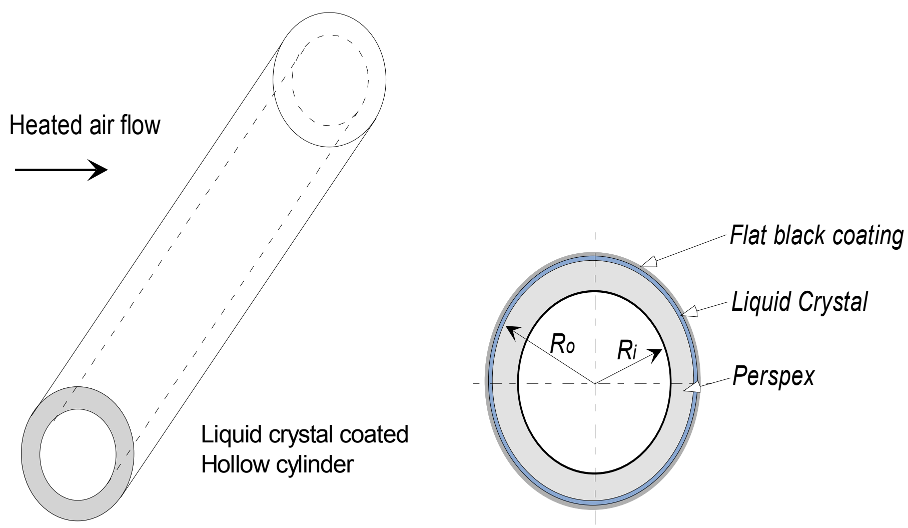
$$J = St\ Pr^{\frac{2}{3}} = \left(\frac{\overline{Nu}}{Re_{Dh}\ Pr}\right) Pr^{\frac{2}{3}}$$

Efficiency Index

$$I = J/f$$

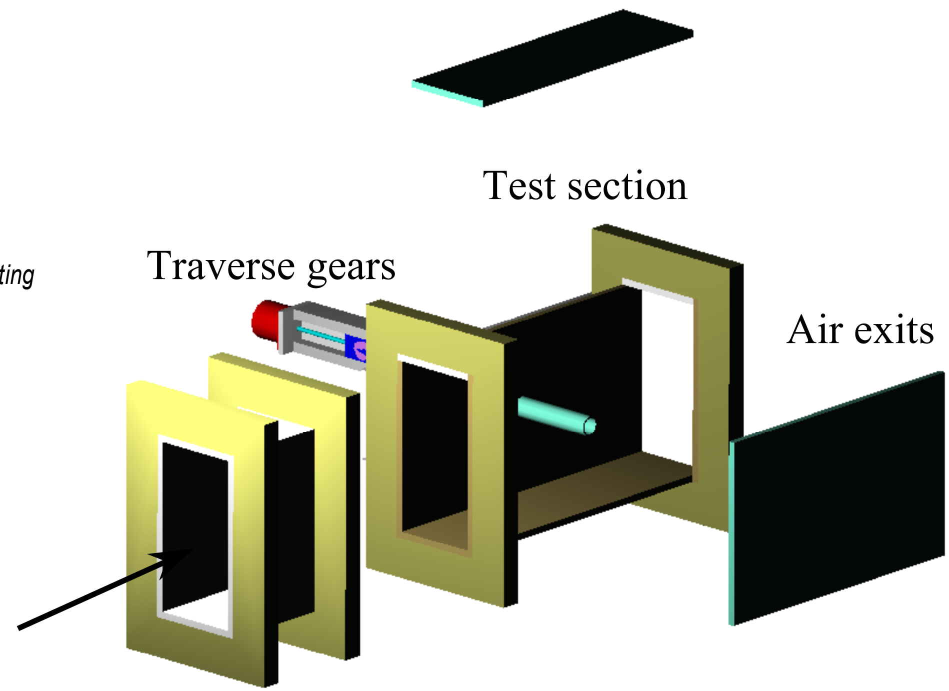
Future Works

(1) Detailed local heat transfer mechanism is anticipated by using Thermochromic Liquid Crystal (TLC) / Infra Red imagings.



Colour change of TLC is monitored from inside of the cylinder

(2) Construct heat sink design parameters using LFMs e.g. cell size effects, number of stacked layers, aspect ratio of the heat sink channels etc.



Test rig for single cylinder and a different version of rig for LFMs will be constructed.

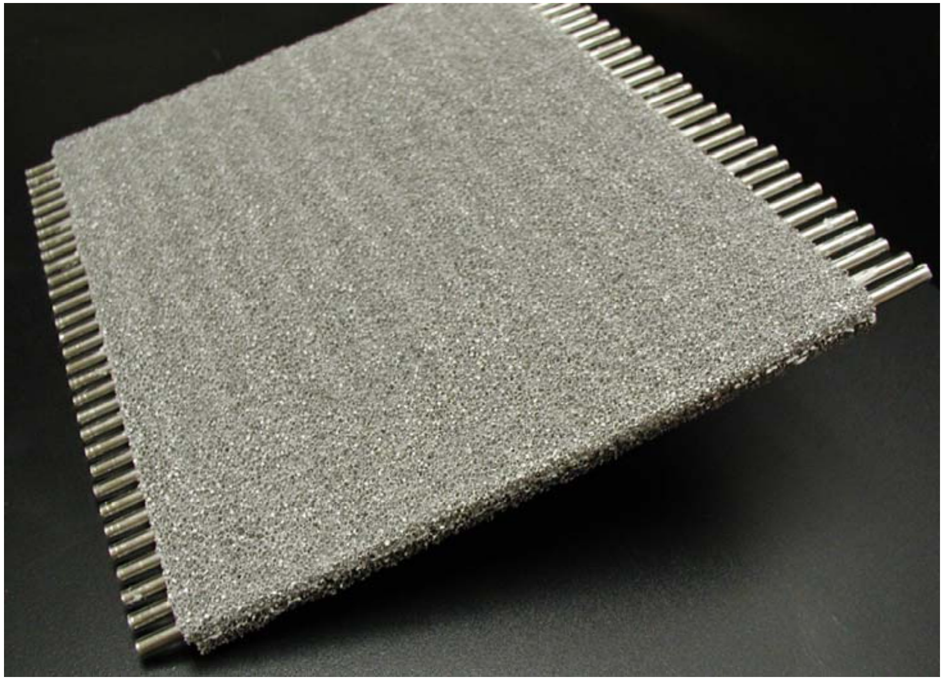
Light weight metal foam heat exchangers

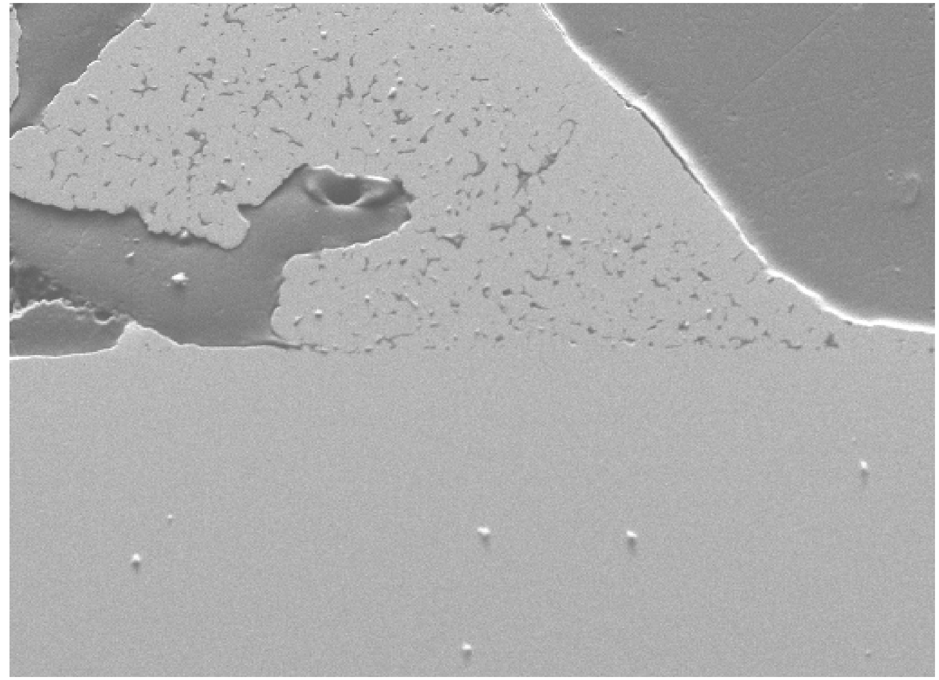
-Examples for heat exchanger application

-Flow resistance

-Thermal resistance

Fabrication of metal foam heat exchangers



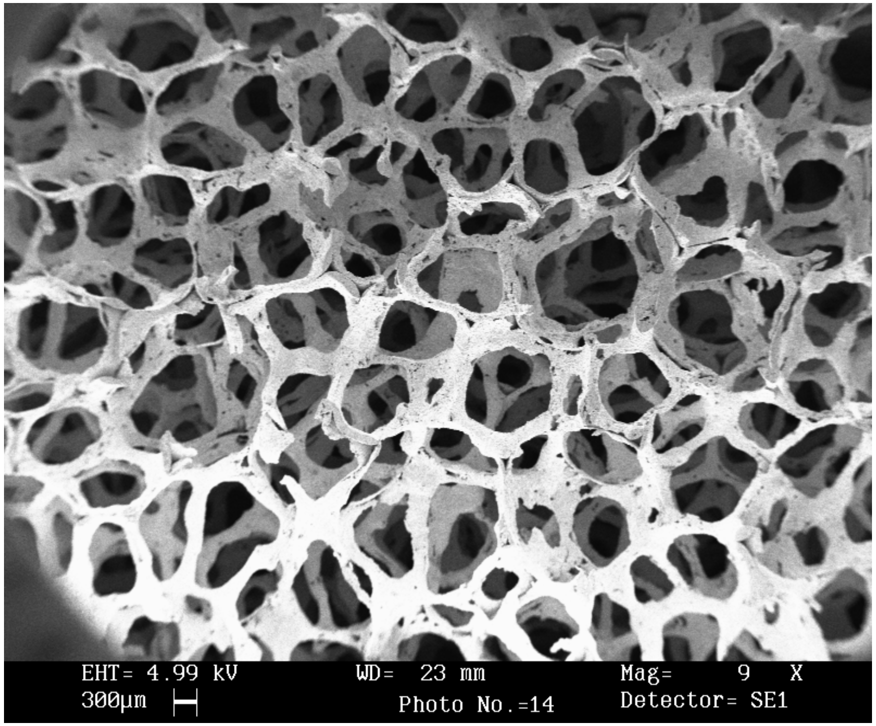


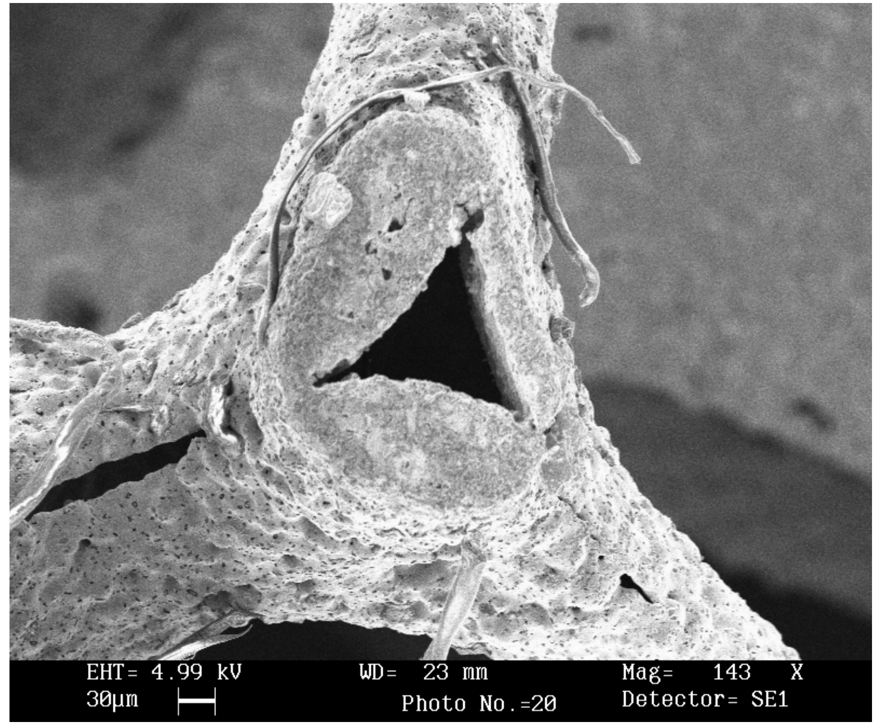
Metal foam compact heat exchanger for high temperature service.
Foam material is PFCT's FeCrAlY.

SEM micrograph of a foam strut sintered to a solid tube.
Bonding region shows metallurgical sintering between foam and solid.

Fabrication of metal foam heat exchangers

Example assemblies manufactured in a proprietary co-sintering technique





SEM images of reticulated metal foam structure (FeCrAlY).
Interconnected tortuouspathways create turbulence
in through-flowing fluids.

Governing Equations

Momentum Equation

$$0 = -\nabla P_f + \frac{\mu_f}{\varepsilon}\nabla^2 u - \frac{\mu_f}{K}u - \frac{\rho_f F}{\sqrt{K}}u^2$$

Energy Equation

$$0 = \frac{\partial}{\partial x}\left(k_{se}\frac{\partial T_s}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_{se}\frac{\partial T_s}{\partial y}\right) - h\tilde{a}\left(\langle T\rangle_s - \langle T\rangle_f\right)$$

$$\rho C_f u\frac{\partial T_f}{\partial x} = \frac{\partial}{\partial x}\left((k_{fe}+k_d)\frac{\partial T_f}{\partial x}\right) + \frac{\partial}{\partial y}\left((k_{fe}+k_d)\frac{\partial T_f}{\partial y}\right) + h\tilde{a}(T_s - T_f)$$

$\tilde{a}$ = wetted area ratio

k_d = thermal dispersion conductivity
$= C_D(\mathrm{Re}_k \Pr_e)u/u_m\, k_e$

$C_D = 0.10$

$\mathrm{Re}_k = \left(u_m\sqrt{K}\right)/\nu$

$\Pr_e = \left(\mu C_f\right)/k_e$

k_e = effective conductivity; $k_{fe} + k_{se}$

F = inertial coefficient

$K = C(1-\varepsilon)^m\left(d_f/d_P\right)^n d_P^2$

where C= 0.00073, m = -0.224, n = -1.11

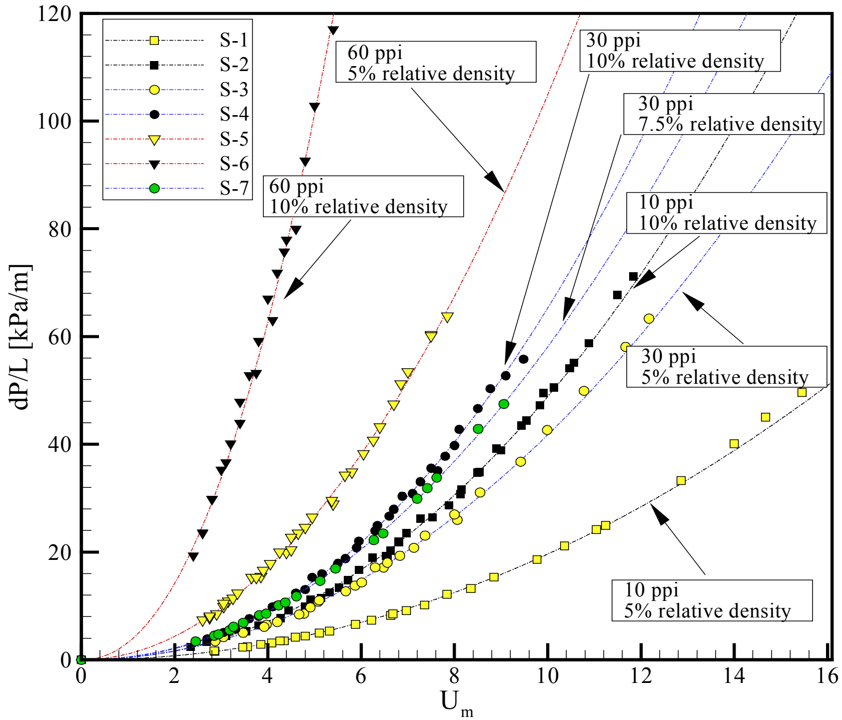
$h = 0.52\,\mathrm{Re}^{0.5}\Pr^{0.37} k_f/d_f$

Fully developed velocity field

Constant heart flux $\quad q = q_w = k_{fe}\left.\frac{\partial T_f}{\partial y}\right|_{y=0} + k_{se}\left.\frac{\partial T_s}{\partial y}\right|_{y=0}$ and $T_s = T_f$ at y = 0

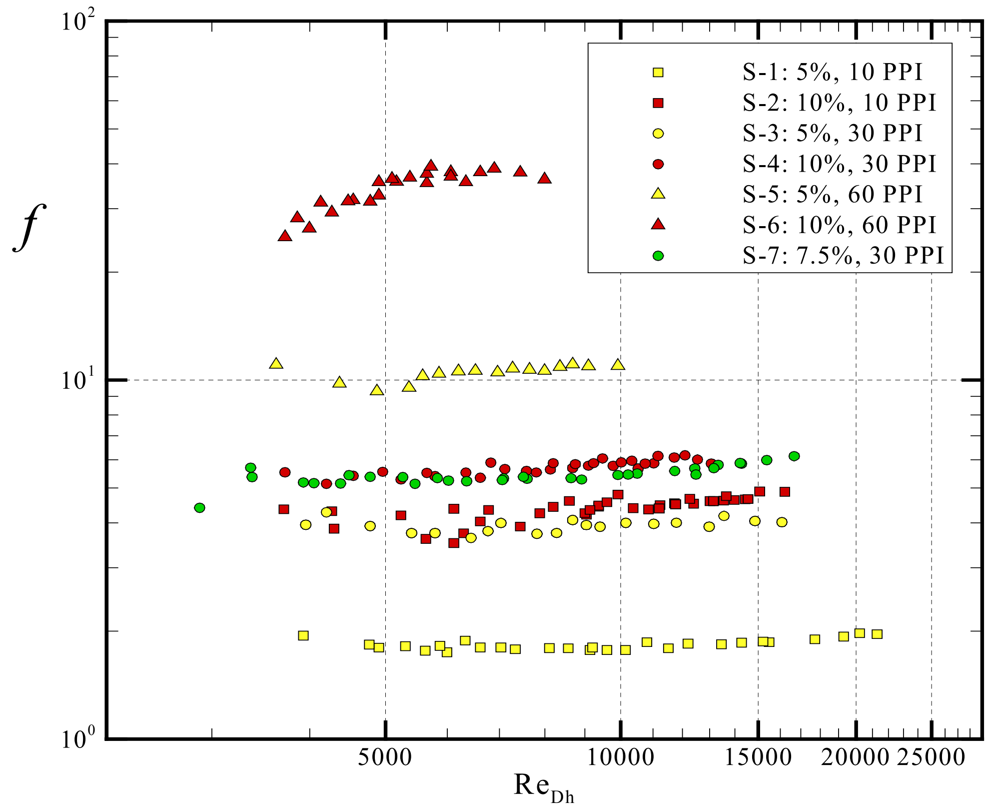
Results of Hydraulic Resistance

- Static pressure drop per unit length



- Friction Factor

$$f = \frac{\Delta P}{L}\frac{D_h}{2\rho\, U_{in}^2}$$

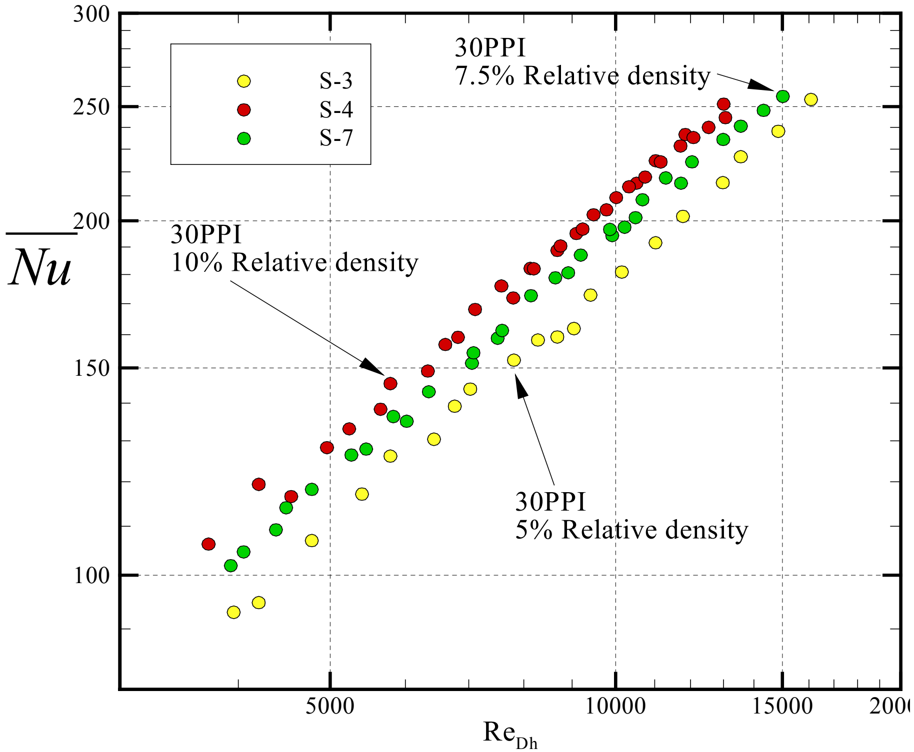


Operating range $Re_{D_h} = \rho\, UD_h / \mu$

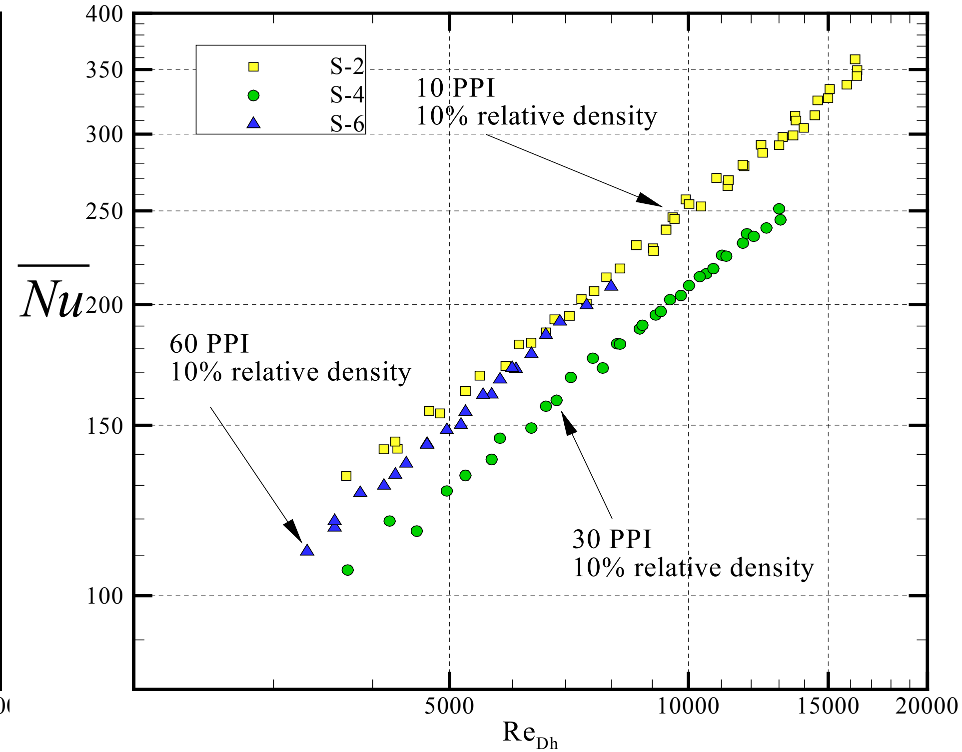
$3000 \leq Re_{D_h} \leq 22000$

Average Nusselt number,

Case 1. fixed pore size as 30 PPI



Case 2. fixed relative density as 10%

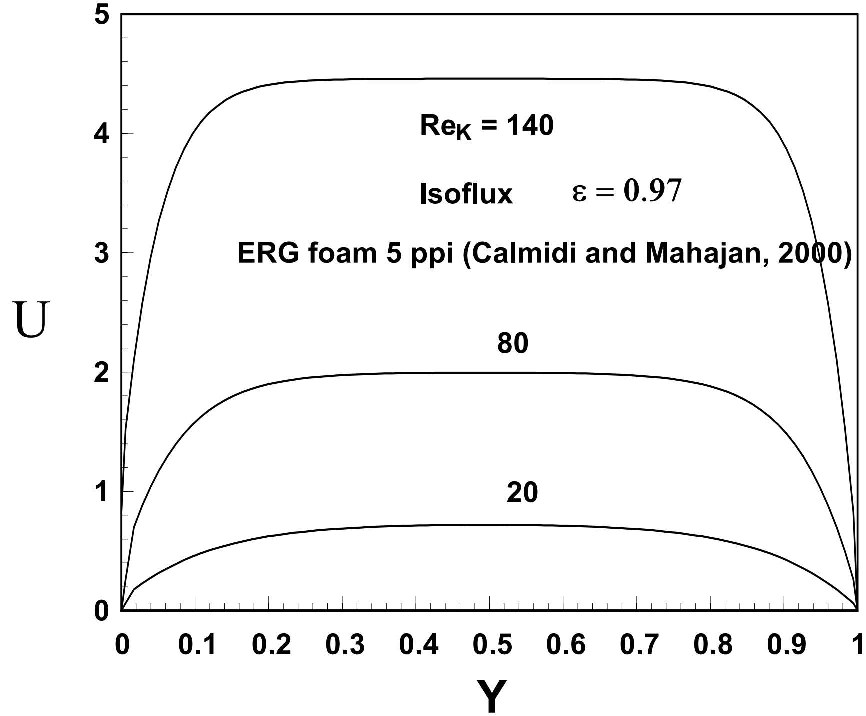


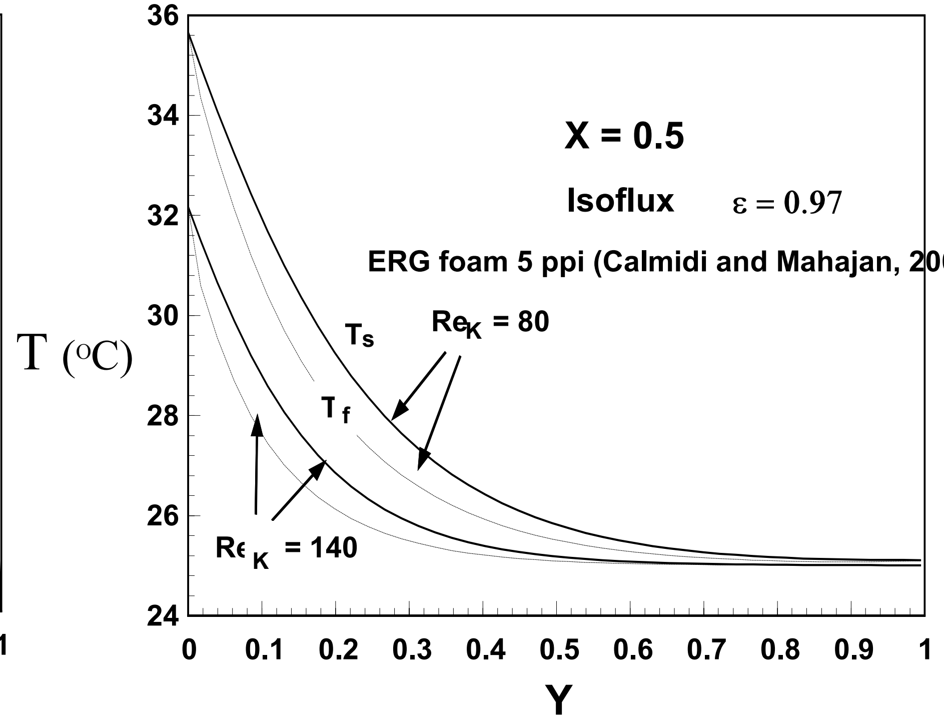
Heat transfer parameters

$$\Delta T_{avg} = \left(\sum_{i=1}^{n} T_{w,i} \right) / n - T_{in}$$

$$\overline{h} = q / \Delta T_{avg}$$ and $$\overline{Nu} = \overline{h} D_h / k_f$$

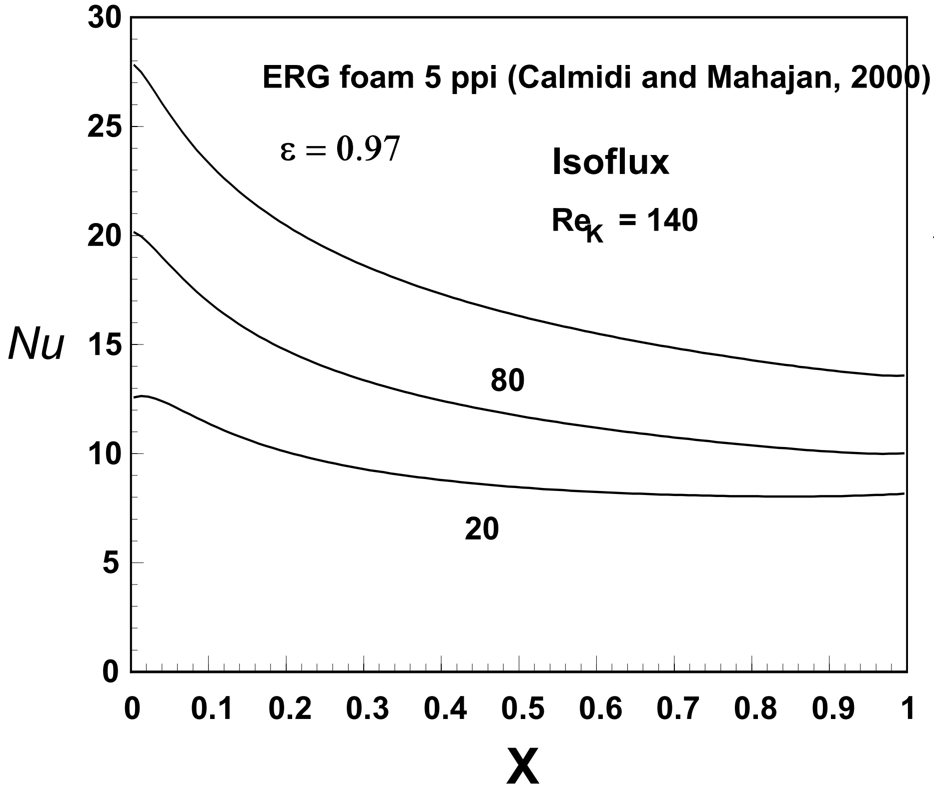
$U = u/u_m \qquad Y = y/H \qquad \mathrm{Re}_K = \left(u_m\sqrt{K}\right)/\nu$

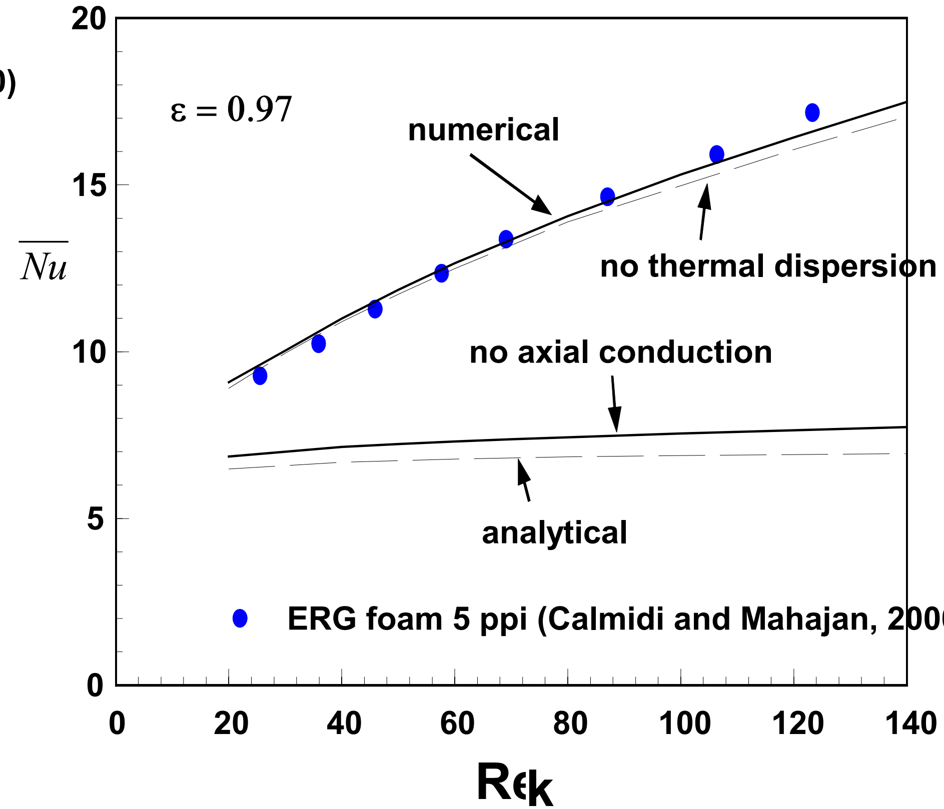




Velocity and temperature distributions for different Reynolds numbers

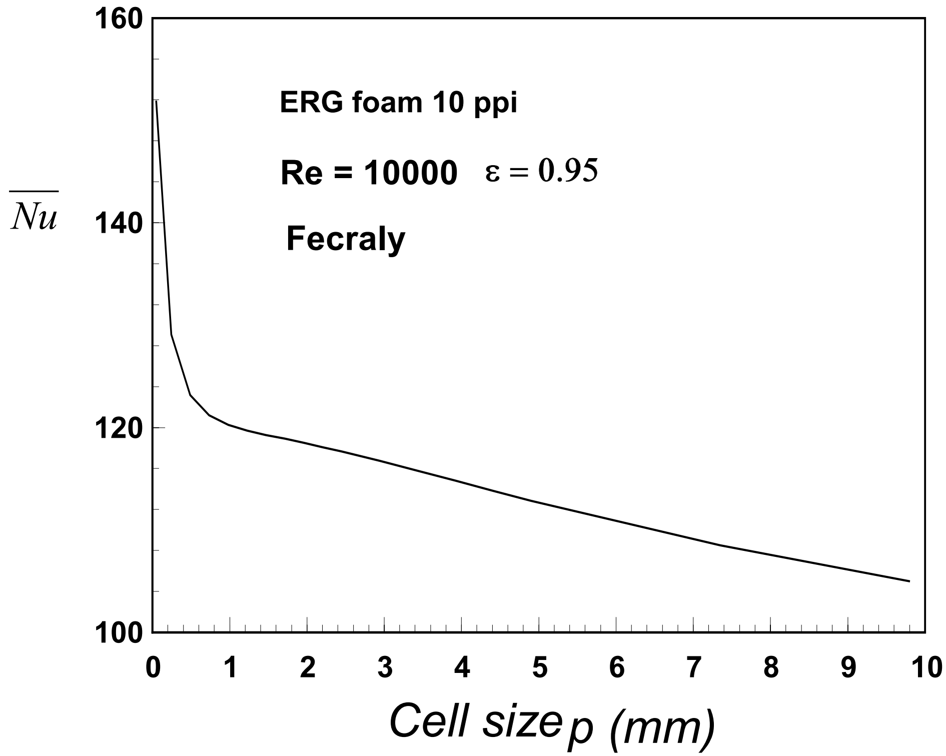
$$Nu(x) = \frac{q}{T_w - T_{f,b}} \frac{L}{k_e} \qquad X = \frac{x}{L} \qquad \overline{Nu} = \frac{1}{L}\int_0^L Nu(x)dx$$

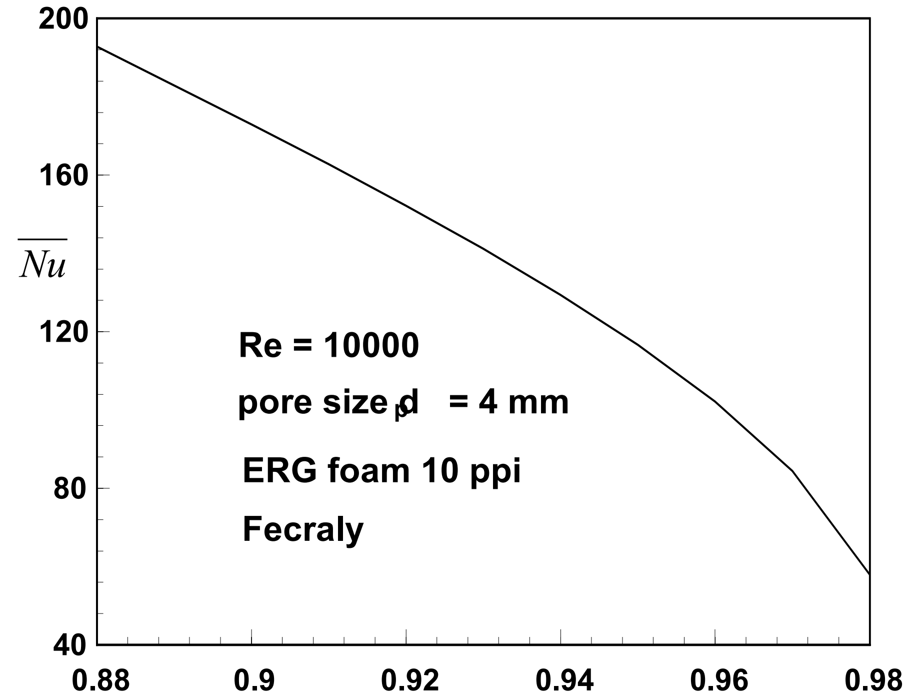




Predictions of local and overall Nusselt numbers

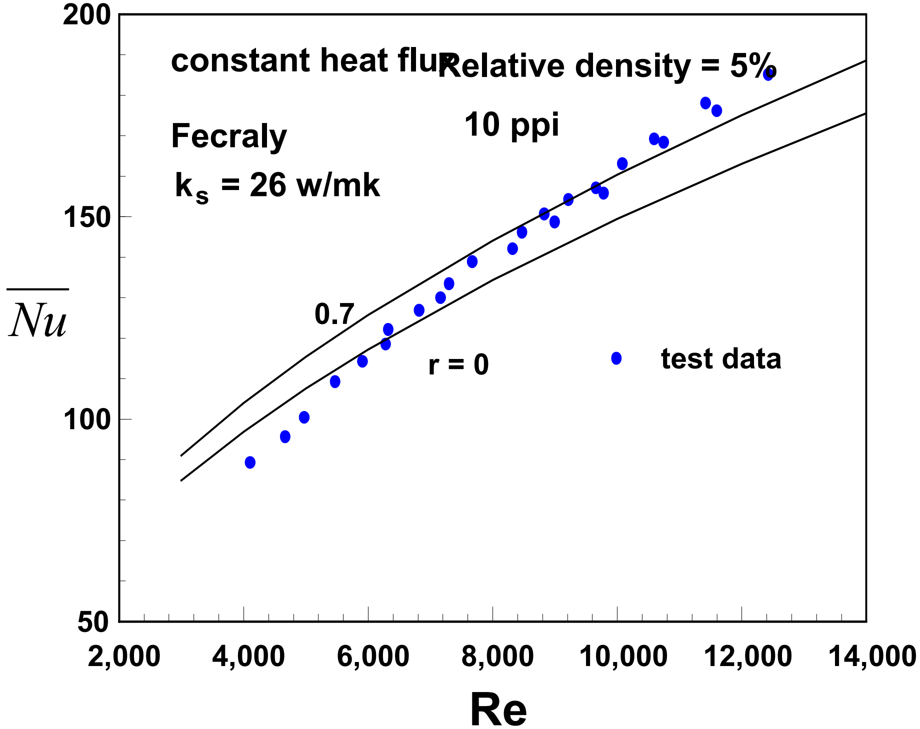
$$\mathrm{Re} = \frac{u_m D_h}{\nu} \qquad Nu(x) = \frac{q}{T_w - T_{f,b}} \frac{D_h}{k_f} \qquad \overline{Nu} = \frac{1}{L}\int_0^L Nu(x)dx$$

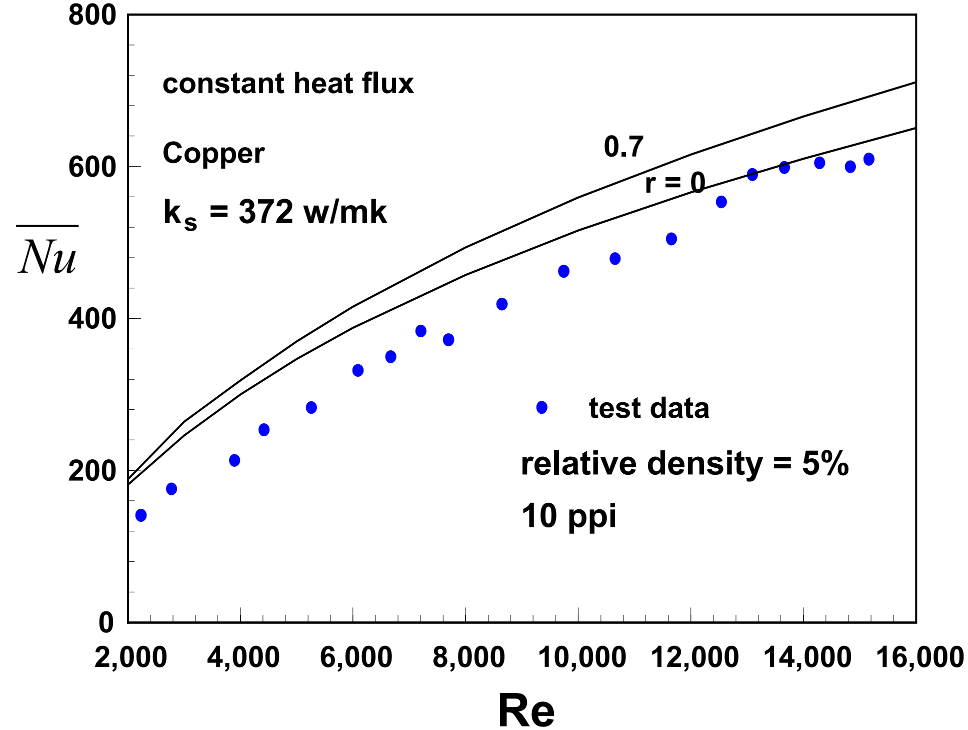




Variations of overall Nusselt number with pore size and porosity

$$Re = \frac{u_m D_h}{\nu} \qquad Nu(x) = \frac{q}{T_w - T_{in}} \frac{D_h}{k_f} \qquad \overline{Nu} = \frac{1}{L}\int_0^L Nu(x)dx$$

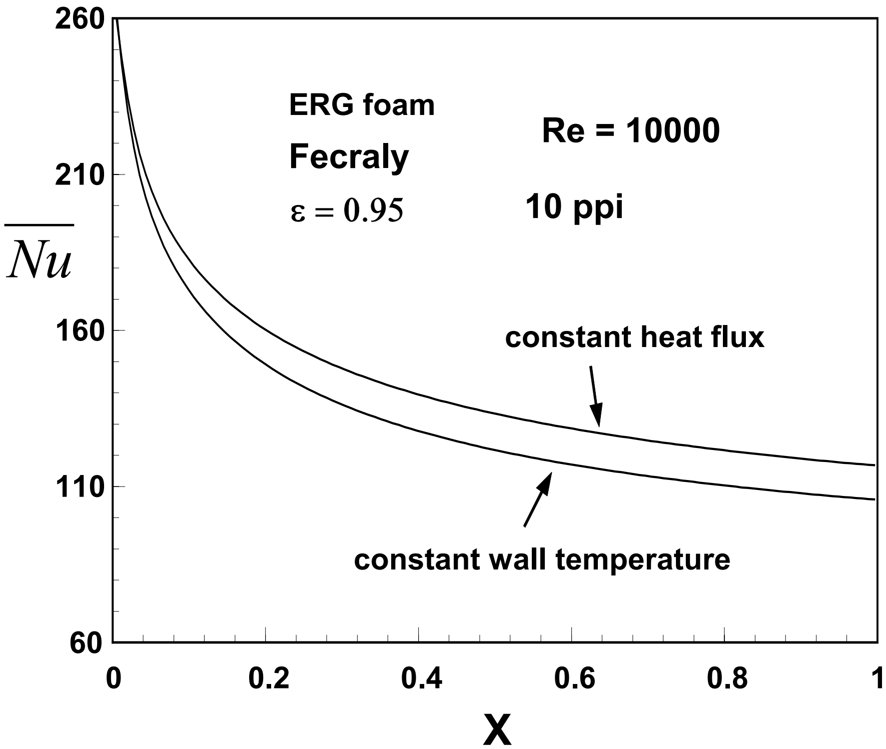


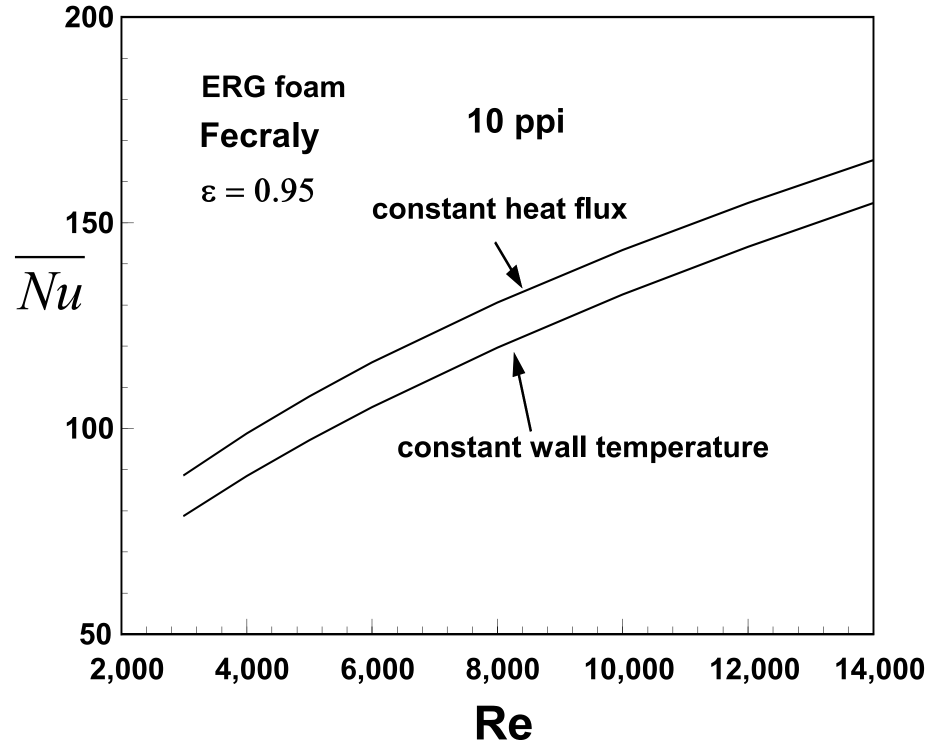


Variations of overall Nusselt numbers with Reynolds numbers for ***Porvair*** material

$$Nu(x) = \frac{q}{T_w - T_{f,b}} \frac{D_h}{k_f}$$

$$\overline{Nu} = \frac{1}{L}\int_0^L Nu(x)dx$$

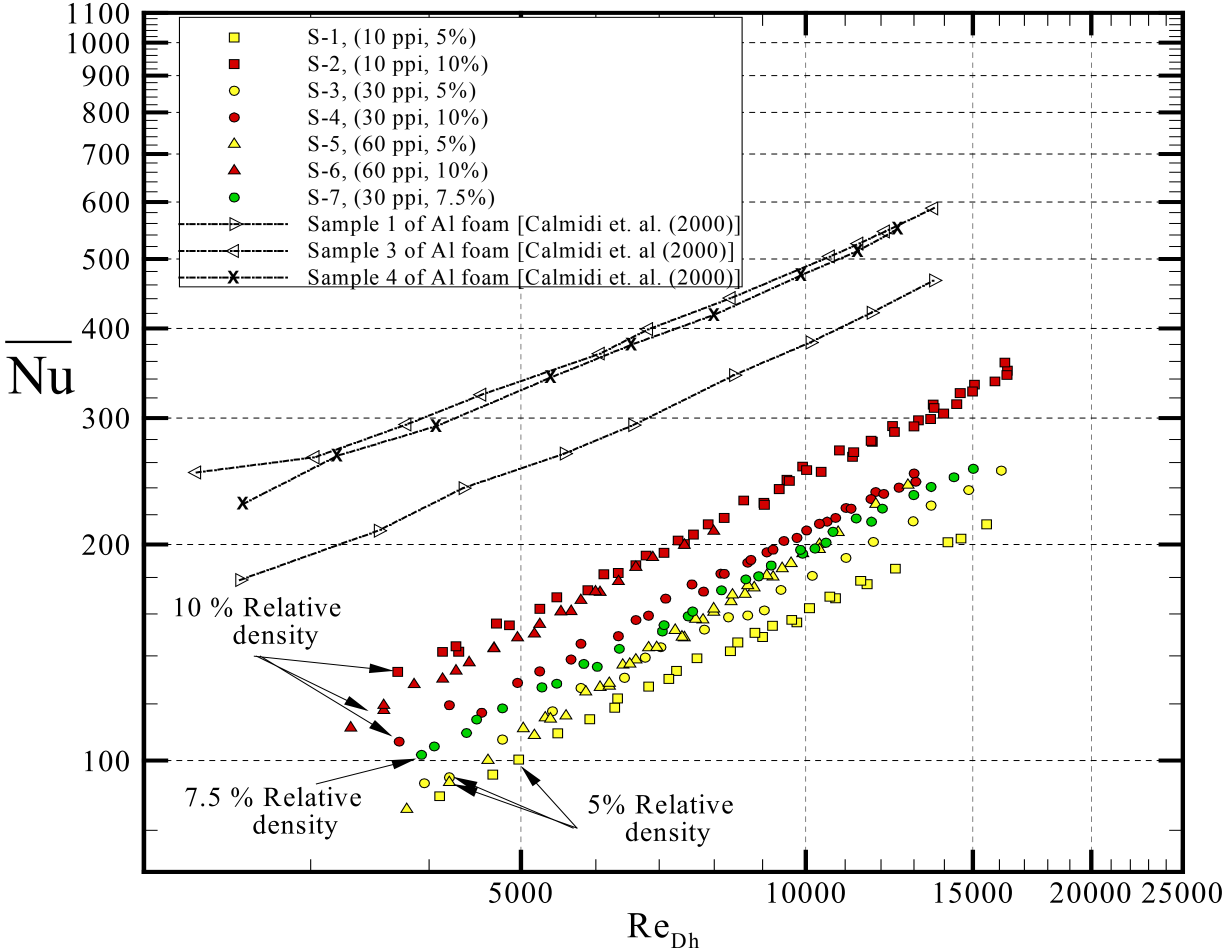




Boundary effects on local and overall Nusselt numbers

Comparisons with solid strut metal foams

[from Calmidi et al. (2000)]



Thermal conductivity

k_s = 16 W/mK (FeCrAlY)

~ 160 W/mK (Al alloy, T-6201)

Nusselt number based on hydraulic diameter

$$\overline{Nu} = \bar{h}D_h / k_f$$

where red, green and yellow filled symbols indicate 10%, 7.5% and 5% relative density, respectively